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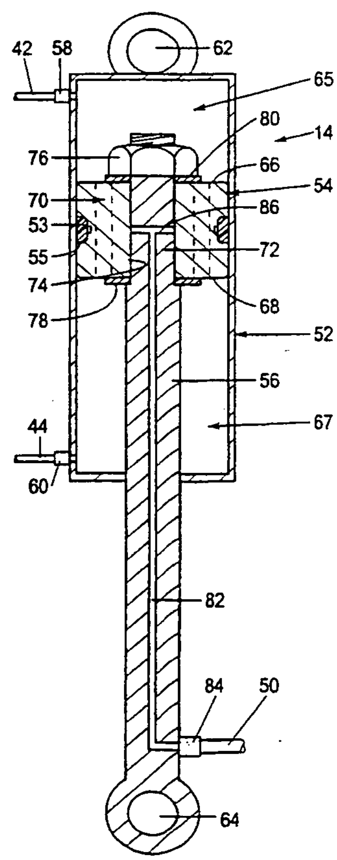
(54) Title: ACTIVE SUSPENSION SYSTEM AND COMPONENTS THEREFOR

(57) Abstract

An active suspension system for a vehicle including at least one fluid operated ram (14) having a cylinder (52) and a main piston (54) mounted therein for reciprocating movement, control means for controlling an equilibrium position of the piston (54) in the ram (14) in response to lateral acceleration of the vehicle in order to counter the effects of body roll of the vehicle, and wherein the ram (14) includes shock absorbing means (70) for permitting rapid movement of the piston (54) from the equilibrium position when operating fluid in the ram (14) is subjected to transient increases in pressure. For each ram (14), first and second working chambers (65, 67) are defined in the cylinder (52) on opposite sides of the piston (54) and the piston (54) includes at least one damping passage therethrough and valve means to control flow of operating fluid through said damping passage when the fluid in one or the other of the working chambers (65, 67) is subjected to transient increases in pressure.



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ACTIVE SUSPENSION SYSTEM AND COMPONENTS THEREFOR

This invention relates to an active suspension system and components therefor including a fluid operated ram.

Active suspension systems are known. Generally speaking the object of such systems is to counteract the effect of body roll which is caused by lateral acceleration when cornering.

An object of the invention is to provide an improved active suspension system for a vehicle.

Another object of the invention is to provide a fluid operated ram for use in an active control system for a vehicle.

Another object of the invention is to provide a shock absorber having adjustable stiffness.

Another object of the invention is to provide a fluid operated ram which is capable of supporting a load by having a pressure differential across its piston and which enables damping movements of the piston to take place in response to transient increases in pressure whilst still being able to support the load.

According to the present invention there is provided an active suspension system for a vehicle including at least one fluid operated ram having a cylinder and a main piston mounted therein for reciprocating movement, control means for controlling an equilibrium position of the piston in the ram in response to lateral acceleration of the vehicle in order to counter the effects of body roll of the vehicle, and wherein the ram includes shock absorbing means for permitting rapid movement of the piston from said equilibrium position when operating fluid in the ram is subjected to transient increases in pressure.

The invention also provides an hydraulic ram which can be used as an adjustable shock absorber, said ram having a cylinder, a main piston mounted on a piston rod and defining first and second working chambers in the cylinder;

at least one first damping passage extending through the main piston and having a first damping valve therein for permitting flow of operating fluid from the first working chamber to the second working chamber;

at least one second damping passage extending through the main piston and having a second damping valve therein for permitting flow of operating fluid from the second working chamber to the first working chamber;

first and second transient pressure relief valves in said first and second damping passages respectively;

the first transient relief valve including a first transient relief piston mounted for sliding movement in a first bore in said main piston, said first relief piston having a first end which is in fluid communication with said first working chamber;

the second transient relief valve including a second transient relief piston mounted for sliding movement in a second bore in said main piston, said second relief piston having a first end which is in fluid communication with the second working chamber; and

a control fluid duct in said piston rod in communication with passages in the main piston to provide fluid communication between said fluid control duct and said first and second bores whereby second ends of the first and second transient relief pistons are subjected to the fluid pressure applied to said fluid control duct, and wherein the arrangement is such that hydraulic fluid supply means can be used to supply operating fluid at a high pressure and a low pressure to said first and second working chambers whereby the pressure difference across the main piston enables the ram to support a load and wherein the high pressure is, in use, applied to said control fluid duct so that the first and second transient relief pistons are closed but, if a transient increase of fluid pressure is caused in the first or second working chamber, the first or second transient relief pistons will be transiently displaced to permit flow of fluid through the first or second damping passage respectively.

The invention also provides a vehicle incorporating the active suspension defined above.

The invention also provides an active suspension system for a vehicle including at least one fluid operated ram, control means for controlling the ram to counter the effects of body roll of the vehicle characterised in that the ram includes shock absorbing means for permitting rapid movement thereof when subjected to transient increases in operating fluid within the ram.

Preferably the ram includes a cylinder and a piston and wherein the control means operates to cause the piston to move to an equilibrium position relative to the ram, which equilibrium position is dependent upon lateral acceleration of the vehicle.

Preferably further, the shock absorbing means permits rapid movement or oscillation of the piston about the equilibrium position so that the ram functions as a shock absorber.

Preferably further, the shock absorbing means includes pressure relief means for relieving pressure spikes in the operating fluid thereof.

Preferably further, the pressure relief means includes an auxiliary piston mounted in a bore formed in the piston (hereinbelow referred to as the main piston) said auxiliary piston being movable when subjected to pressure spikes in said operating fluid to thereby open a pressure relief passage to permit flow of operating fluid through said passage thereby enabling movement of the main piston to counter the shock effect of said pressure spikes.

Preferably further, the main piston is mounted on a piston rod which includes a control fluid passage which passage is coupled to receive operating fluid from the control means and wherein the main piston includes ducts which open to the bore therein wherein one side of the auxiliary piston is subjected to pressure determined by said control means

and the other side of the auxiliary piston is subjected to the same pressure but having superimposed thereon pressure spikes transmitted thereto by the wheel of a vehicle as it traverses rough or uneven surfaces.

The invention also provides a fluid operated ram for use in an active suspension system for a vehicle, said ram including a cylinder, a main piston mounted on the piston rod for reciprocating movement in the cylinder, pressure relief means for relieving, in use, pressure spikes in said operating fluid in the cylinder.

The invention also provides an adjustable shock absorber including a fluid operated ram which includes a cylinder, a main piston mounted on a piston rod for reciprocating movement in the cylinder, a fluid flow passage in fluid communication with said cylinder at first and second points on opposite sides of said piston, pressure relief means for relieving, in use, pressure spikes in operating fluid in the cylinder, and control means operable on said pressure relief means to control response thereof to said pressure spikes to thereby effectively control the stiffness of the shock absorber.

Preferably further, the pressure relief means includes an auxiliary piston mounted for reciprocating movement in said piston and wherein one face of the auxiliary piston is subjected to pressure in operating fluid in the piston and the other face of the auxiliary piston is subjected to a back pressure controlled by said control means.

The invention will now be further described with reference to the accompanying drawings, in which:

FIGURE 1 is a schematic view of the vehicle turning on a circular path;

FIGURE 2 is a schematic view of a front view of the vehicle;

FIGURE 3 is a schematic view of part of an active suspension system of the invention;

FIGURE 4 is a schematic view of the active suspension system of the invention;

FIGURE 5 is a simplified cross-sectional view through a ram of the invention;

FIGURE 6 is a more detailed view in partial axial section:

FIGURES 7, 8 and 9 are schematic representations which illustrate the manner in which the ram of the invention functions:

FIGURES 10 and 11 are isometric views of the main piston:

FIGURE 12 is a side view of the main piston;

FIGURE 13 is a plan view of the main piston;

FIGURE 14 is an underside view of the main piston;

FIGURE 15 is a cross-sectional view along the line 15-15;

FIGURE 16 is a cross-sectional view along the line 16-16;

FIGURE 17 is a cross-sectional view along the line 17-17;

FIGURE 18 is an underside view of the main piston;

FIGURE 19 is a cross-sectional view along the line 19-19;

FIGURE 20 is a cross-sectional view along the line 20-20;

FIGURE 21 is a cross-sectional view along the line 21-21;

FIGURE 22 is a cross-sectional view along the line 22-22;

FIGURE 23 is a cross-sectional view along the line 23-23;

FIGURE 24 is a cross-sectional view along the line 24-24;

FIGURE 25 is a cross-sectional view along the line 25-25;

FIGURE 26 is a schematic view showing in more detail an upper retaining washer;

FIGURE 27 is a schematic view showing in more detail a lower retaining washer; and

FIGURE 28 schematically illustrates an alternative embodiment.

Figures 1 and 2 show a vehicle 2 travelling in a circular path 4 of radius R. The vehicle will be subjected to a centrifugal acceleration A which is proportional to V^2/R where V is velocity of vehicle. The acceleration tends to rotate the vehicle about an axis 6 which is generally horizontal and tangential to the circular path 4, the inner side 8 of the vehicle tending to lift whereas the outer side 10 of the vehicle tending to move closer to the road 3. In accordance with the invention, the vehicle 2 has an active suspension system 12 which tends to counteract the body roll. Thus, on cornering, the active suspension system

12 tends to raise the side of the vehicle on the outside of the turn and lower the vehicle on the inner side of the turn, as will be explained hereinafter.

Figures 3 and 4 are schematic views which illustrate the principles of the active suspension system of the invention. The system includes four double-acting hydraulic rams 14, 16, 18 and 20. Each of these rams is associated with one of the wheels (not shown) of the vehicle 2. The rams 14 and 16 are associated with the front and rear wheels on the right hand side of the vehicle and the rams 18 and 20 are associated with the front and rear wheels on the left hand side of the vehicle. The rams 14, 16, 18 and 20 replace shock absorbers which would normally be provided in a vehicle suspension. Figure 3 diagrammatically shows the ram 14 connected between the vehicle chassis 22 and an axle assembly 24 which includes an axle 26 upon which the front right hand side wheel of the vehicle is mounted. The axle assembly 24 is coupled to a drop arm 28, one end of which is typically pivotally connected to the chassis 22 of the vehicle. A coil spring 30 acts between the chassis 22 and the axle assembly 24, in the usual way. Conveniently the ram 14 is located within the coil spring 30. The other rams can be mounted in the same or similar way.

Figure 4 schematically illustrates the hydraulic control system for operating the rams 14, 16, 18 and 20. The system includes an hydraulic pump 32 having an outlet line 34 which is coupled to an inlet of a control valve 36. The control valve 36 is coupled to a pendulum which includes a pendulum mass 38 and pendulum arm 40. The control valve 36 can be identical to that disclosed in WO 94/02767 and the content of that publication is incorporated herein by reference. The control valve 36 has outlet lines 42 and 44 and a return line 46 which may include an hydraulic fluid tank (not shown). The control valve 36 is mounted on the vehicle such that the axis 48 of the pendulum arm 40 is oriented in the longitudinal direction of the vehicle (i.e. parallel to the axis 6 shown in Figure 1). When the vehicle turns, the pendulum arm will swing to the left or right according to the direction of the turn. The control valve 36 causes the pump 32 to produce a relatively high pressure on its outlet line 34 which is also present at one of the outlets of the valve 36, the other outlet remaining at the relatively low return pressure of the return line 46 to the

pump. Generally speaking, hydraulic lines are connected from the outlets of the control valve 36 to the rams 14, 16, 18 and 20 but in an opposite configuration on the left hand and right hand sides of the vehicle. More particularly, the outlet line 42 is connected to the tops of the rams 14 and 16 but to the bottoms of the rams 18 and 20. Similarly, the outlet line 44 is coupled to the tops of the rams 18 and 20 but to the bottoms of the rams 14 and 16. The system includes a control line 50 which is connected from the outlet line 34 of the pump to pistons of the rams as will be described in more detail below.

The operation of the active control suspension is as follows. When the vehicle 2 is moving in a straight line, the valve 36 effectively causes the outlet line 34 of the pump to be coupled to the return line 46. Thus the outlet pressure of the pump is relatively low, say 15psi. When, however, the vehicle turns left, the pendulum operates on the valve 36 so as to cause an increase in pressure at outlet 42 whilst the pressure at outlet 44 remains at the pump return pressure. This causes the rams 14 and 16 on the right hand side of the vehicle to elongate and the rams 18 and 20 on the left hand side of the vehicle to contract. Because of the coupling of the rams between the axle assemblies and the chassis, the right hand side of the vehicle is raised and the left hand side of the vehicle is lowered thereby countering the effect of body roll.

The valve 36 produces output pressure on its outlet 42, the magnitude of which is generally proportional to the lateral acceleration of the vehicle as described in WO 94/02767 referred to above. This causes the magnitude of the raising and lowering of the body of the vehicle to be proportional to angular velocity as required. In practice, the rams will move to an equilibrium position which is determined chiefly by the magnitude of the lateral acceleration, the weight of the vehicle and the characteristics of the springs 30.

The opposite happens when the vehicle turns towards the right. In this case the control valve 36 operates to elongate the rams 18 and 20 on the left hand side of the vehicle and contract the rams 14 and 16 on the right hand side of the vehicle. Again the degree of raising and lowering is proportional to the lateral acceleration.

Whilst the active suspension system described above is effective for countering the effects of body roll, it will be appreciated that the rams 14, 16, 18 and 20 would, unless special provision is made, effectively mean that there would not be any shock absorbing capacity in the suspension. This is basically unacceptable for passenger or other vehicles and therefore, in accordance with the invention, special provision is made to enable the rams 14 and 16 to operate in a dual way. That is to say, in a first mode of operation they operate as normal double acting hydraulic rams which will move to equilibrium positions which are determined by lateral acceleration and other factors, as mentioned above, in order to counter the effects of body roll of the vehicle and, in a second mode, in which they are able to accommodate transient displacements from the equilibrium position in order to function as shock absorbers which would normally be incorporated into a vehicle suspension system. It will be appreciated that whilst the rams are operating as shock absorbers, they must still be generally capable of supporting the loading applied thereto otherwise the suspension of the vehicle would not function satisfactorily. In other words if the rams did not maintain the load applied thereto, the wheels of the vehicle would collapse into the wheel arches or move to fully extended positions which of course is unacceptable. The transient operation of the rams will be described below.

Figures 5 and 6 illustrate in more detail the structure of the ram 14 which enables the transient response. The rams 16, 18 and 20 can be of similar construction and therefore need not be described.

The ram 14 schematically shown in Figure 5 includes a cylinder 52 and a main piston 54 which is mounted on a piston rod 56, the piston and piston rod being slidable within the cylinder 52, in the usual way. A seal 53 is located in a groove 55 extending about the circumference of the piston, as shown. Fluid couplings 58 and 60 are provided near the top and bottom of the cylinder 52. These couplings enable fluid communication with the valve 36. For the ram 14, the upper coupling 58 is coupled to the outlet line 42 and the lower coupling 60 is coupled to the outlet line 44. The top of the cylinder 52 is provided with a coupling 62 which enables it to be connected to the chassis 22 of the vehicle, in the usual way. A coupling 64 is provided on the lower end of the rod 56 to

enable the lower end of the shaft to be coupled to the axle assembly 24. The ram 14 can alternatively be mounted in an opposite position to that shown in Figure 3, in other words, with the cylinder 52 connected to the axle assembly and the piston rod 56 connected to the vehicle chassis 22. Upper and lower working chambers 65 and 67 are defined within the cylinder 52 above and below the main piston 54.

When the vehicle is stationary or moving in a straight line, the pressure in the lines 42 and 44 is the same and the piston 54 is arranged to be in an equilibrium position which is generally centrally located relative to the cylinder 52. The vehicle suspension and springs dictate the normal static position of the vehicle body (sprung mass) relative to the wheels and axles (unsprung mass), in the usual way. If the vehicle turns left, the pressure in line 42 will increase in accordance with the magnitude of the lateral acceleration but the pressure in line 44 will remain at the low return pressure level. This causes the piston 54 to move downwardly to occupy a new equilibrium position which, as described above, will be dependent on a number of factors including the magnitude of the lateral acceleration, weight of the vehicle and characteristics of other components of the vehicle's suspension including the coil spring 30. The control valve will maintain the pressure increase in the line 32 as long as the lateral acceleration of the vehicle continues. Accordingly, there will be pressure differentials across the pistons in the rams so that the rams can support different loads during cornering. In accordance with the invention, the ram 14 includes shock absorbing means for enabling transient or rapid movement of the piston 54 in the cylinder in order to smooth out the effects of shock increases in pressure in the operating fluid within the cylinder caused by perturbations of the road over which the wheel of the vehicle travels. As explained below, the shock increases in pressure are damped, in accordance with the invention, in a way which still enables the pressure differentials to be maintained across the pistons in the rams so that the rams support the loads applied thereto.

If, for example, the wheel of the vehicle strikes a bump or upward projection, the wheel will cause a transient upward force to be transmitted through the piston rod 56. The upper face 66 of the piston 54 will tend to compress the operating fluid in the cylinder above the piston causing a pressure spike therein. Unless this is relieved, the suspension

would be very stiff and unpleasant for passengers in the vehicle. If the wheel of the vehicle encounters a pothole or the like, the springs 30 will tend to cause a rapid rebound or downward movement of the wheel which has the effect of causing the lower face 68 of the piston 54 to produce a pressure spike in the fluid in the cylinder beneath the piston. Again, unless this is relieved, the suspension would not exhibit shock absorbing capacity. In accordance with the invention, the piston 54 includes pressure relief means 70 which is responsive to pressure spikes in the fluid in the cylinder 52 on either side of the piston 54 and enables flow of hydraulic fluid therethrough in order to allow the piston 54 to move or oscillate so as to negate the effects of these pressure spikes. The piston 54 will return to its equilibrium position prior to the pressure spike which is appropriate for moving in a straight line or cornering left or right, as the case may be. When the vehicle is cornering, there will be a pressure differential in the working chambers 65 and 67 but when the vehicle is moving in a straight line, there will be no differential across the piston 54.

In the illustrated arrangement, the upper end of the rod 56 has a shoulder which defines a narrower diameter portion 72 which extends through a central axial bore 74 of the piston 54. The upper end of the narrower diameter portion 72 is threaded and a retaining nut is provided to clamp the piston 54 against the shoulder in the rod 56. A lower washer 78 is located between the shoulder in the rod 56 and the piston 54. An upper washer 80 is located between the nut 76 and the upper face 66 of the piston.

The rod 56 includes an axially extending control duct 82, the lower end of which is in fluid communication with a coupling 84 for connection to the control line 50 of the pump 32. The control duct 82 communicates with radially extending ducts 86. As best seen in Figure 6, there is at least one duct 88 which extends from the duct 86 to the groove 55 so that hydraulic fluid pressure in the duct 86 will bias the seal outwardly into sealing engagement with the inner surface of the cylinder 52.

Figure 6 diagrammatically shows the pressure relief means 70 in more detail. Figure 6 is a fragmentary axial cross-section through the ram 14 constructed in accordance with the invention. It will be seen that the piston 54 includes two valve bores 92 and 94.

Upper and lower pressure relief valve pistons 96 and 98 are slidably mounted in the bore 92. A compression spring 100 acts between the opposed inner faces of the valve pistons 96 and 98. The outer end faces 102 and 104 of the valve pistons 96 and 98 are exposed to the operating fluid within the cylinder 52. The spring 100 biases the pistons 96 and 98 apart and into engagement with the washers 80 and 78 respectively. The washers 80 and 78 only partly cover the bores 92 and 94 so that hydraulic fluid can enter these bores, as will be described in more detail below. The bore 92 also includes upper and lower pressure relief valve pistons 106 and 108 which are biased apart by means of a compression spring 110. Again, the end faces 112 and 114 are engaged by the washers 80 and 78 so as to retain the valve pistons in the bore 94. The ducts 86 from the control duct 82 communicates with the valve bores 92 and 94. Accordingly, the fluid pressure in the control duct 82 is always uniformly applied to the inner faces of all four valve pistons 96, 98, 106 and 108. The function of the springs 100 and 110 is essentially to always ensure that the duct 86 is not blocked by movement of one or more of the valve pistons. The valve pistons 96 and 98 effectively seal against the valve bore 92 and similarly the valve pistons 106 and 108 seal against the valve bore 94. Accordingly, when the valve pistons 98 are in their closed positions, as shown in Figure 6, the main piston 54 acts like a solid body and the ram 14 can function in the usual way. Its movement can therefore be controlled by pressure differences between fluid in the lines 42 and 44.

The piston 54 includes two (or more) damping ducts 116 and 118. As will be described in more detail below, hydraulic fluid is permitted to rapidly pass into one or other of the damping ducts in response to a spike or transient increase of pressure above or below the piston. Pressure spikes can be caused by the wheel of the vehicle hitting a bump or falling into a sudden depression. Normally the spikes would be damped out by means of a shock absorber and the damping ducts 116 and 118 enable controlled damping to take place in the ram of the invention.

Figures 7 to 9 are schematic illustrations of the way in which the valve pistons and damping pistons function in the ram 14. In this arrangement, one end of the damping duct 116 opens to the working chamber 67 beneath the lower face 68 of the piston 54 whereas

the damping duct 118 opens to the working chamber 65 above the upper face 66 of the piston 54. The ram of the invention may include more than two of the damping ducts so as to improve the performance of the ram, as will be described in more detail below.

In the diagrammatic arrangement shown in Figure 7, the damping duct 116 is provided with a damping piston 120 which is mounted for sliding movement in a damping bore 122, which forms part of the damping duct 116. A damping spring 124 acts against the lower face of the damping piston 120 and biases the upper face of the piston into a damping valve seat 126. The lower end of the valve 126 is retained in the bore 122 by means of the washer 78. The valve includes a second damping piston 130 which is mounted for sliding movement in a second damping bore 132. A second damping spring 134 biases the lower end face of the damping piston 130 into a valve seat 136. Again, the damping spring 134 is retained in the damping bore 132 by the washer 80.

Figure 7 shows the piston 54 in an equilibrium position. In this position, the pressure on the upper and lower faces 66 and 68 may be the same, in which case the pressure will be at a relatively low level, say of the order of 15 to 30psi or, if the vehicle is cornering, there may be a substantial pressure differential in the working chamber 65 and 67 in order that the ram 14 can support increased loading required for countering the effects of body roll during cornering. As explained above, the control valve 36 is arranged to always supply the higher of the pressures applied to the working chamber 65 and 67 to the control duct 82 and therefore the pressure within with valve bores 92 and 94 is always equal to the pressure which is highest in the working chamber 65 or 67. In the case where the vehicle is moving in a straight line, relatively low values of pressure will be applied to the opposed end faces of each of the pressure relief valve pistons, 96, 98, 106 and 108 and the pistons will remain in their closed positions, as shown in Figure 7. During cornering, even when a substantial pressure differential, say of the order of 500psi or more, is required between the working chamber 65 and 67, the pressure relief pistons will remain in their closed positions because the higher pressure is present in the bores 92 and 94. When the pressure relief valve pistons 96, 98, 106 and 108 are in their closed positions, as shown in Figure 7, the damping pistons 120 and 130 remain seated against their respective valve

seats 126 and 136 because the pressure relief valves prevent any flow through the damping passages 116 and 118. Also, the springs 124 and 134 bias the damping pistons 120 and 130 into engagement with their respective valve seats.

Figure 8 diagrammatically represents the response of the piston 54 when the piston rod 56 has been moved rapidly upwardly such as may occur when a vehicle hits a bump or the like. The rapid upward movement of the piston rod 56 causes a transient increase in hydraulic fluid pressure in the working chamber 65 above the piston 54. When this occurs, the valve pistons 96 and 106 will move rapidly downwardly because the pressure above the main piston 54 to which the upper faces of the pistons 96 and 98 will be transiently very much higher than the equilibrium pressure to which the lower end faces of these pistons are exposed. The pistons 96 and 106 therefore quickly move downwardly, as indicated by arrows 137. Downward movement of the piston 96 opens a transfer port 138 which causes a rapid increase in pressure within the damping duct 116. The pressure will rapidly build up in the damping duct 116 to the point at which the biasing effect of the damping spring 124 is overcome and the damping piston 120 will move downwardly in the damping bore 122. Once this occurs, this will enable flow of hydraulic fluid from the region above the piston 54 to the region below the piston 54. In other words this permits rapid upward movement of the piston 54 and hence the piston rod 56. This causes a reduction in pressure in the working chamber 65 in order to dissipate the transient increase in pressure. When the pressure in the working chamber 65 equals that in the bore 92, the pressure relief piston 96 will close the port 38 and the damping piston 120 will return to its closed position. At this point, however, the piston is rapidly moving upwardly and, as in all dynamic systems, there is likely to be overshoot which will cause a transient reduction in pressure in the cylinder 52 beneath the piston 54. This transient will also be relieved, as will be described below, until the cylinder again moves to an equilibrium position in a damped movement as required. During the initial pressure relief as described in Figure 8, the valve 108 and damping piston 130 remain in their closed positions. The upper valve piston 106 will move downwardly, like the piston 96, but this may or may not open a damping port, depending on the geometry of the configuration. Significantly, however, the damping duct 118 remains closed to flow of hydraulic fluid therethrough. Also, it will be

appreciated that because of the fact that the higher of the two pressures applied in the working chamber 65 and 67 is always present in the bores 92 and 94, the valve 96 can be transiently displaced form its equilibrium position to enable damping to occur but still enabling a pressure differential to be maintained between the working chambers 65 and 67 in order to enable the ram 14 to support loads applied thereto.

Figure 9 schematically illustrates an arrangement in which there is a transient increase in pressure in the working chamber 67 beneath the piston 54. This can be caused by the wheel of the vehicle falling into a pothole or the like, in which case the springs 30 will cause a rapid downward movement of the piston 54. The same effect can be caused during overshoot of the piston 54 in response to striking a bump or the like. In this case, when there is a transient pressure increase in the hydraulic fluid in the chamber 67, both the valve pistons 98 and 108 will be caused to rapidly move upwardly against their respective springs 100 and 110, as indicated by arrows 142. Upward movement of the piston 108 causes opening of a port 144 which permits flow of hydraulic fluid into the damping duct 118, as indicated by arrow 146. This causes a rapid build-up of pressure within the duct 118 and when the pressure has built-up sufficiently, it will cause the damping piston 130 to move away from the valve seat 136 and cause flow of fluid through the duct 118 into the working chamber 67, as indicated by arrows 146. Again, the compression spring 110 against which the valve piston 108 is quite light and therefore the port 144 is opened very rapidly. The movement of the damping piston 130 is subject to overcoming the biasing force of the damping spring 134 and the spring force can be adjusted so as to affect the dynamics of the response and hence the degree of damping.

It will be appreciated that if the damping springs 124 and 134 are very light, then rapid damping will occur and the suspension will be very "soft". If, however, the spring forces of the springs 124 and 134 is greater, the damping will be less and the suspension will be correspondingly stiffer.

It will be appreciated from Figures 6, 7 and 8 that spikes or transient increases in pressure in the hydraulic fluid above or below the main piston 54 will cause selective

opening of the valve pistons which in turn expose the damping pistons to the transient increases in pressure. The damping pistons will move once the biasing force of the damping compression springs have been overcome. Thus the ram can function as a normal double acting hydraulic ram which incorporates a shock absorbing action. The degree of shock absorbing action can be controlled by a number of factors such as the relative diameters of the pistons, sizes of ports and so on but the main factor will be the biasing force applied by the damping springs 124 and 134. The damping action is operable to damp transient increases in pressure in the working chambers 65 and 67 caused by the wheel of the vehicle hitting a bump or depression and/or by overshoot caused by initial and subsequent responses of the piston to the initial transient. After damped movement of the main piston 54, it will return to its equilibrium position as determined by the pressure generated by the valve 36. As explained above, the valve 36 produces relatively low pressures when the vehicle is moving in a straight line and the low pressures are applied to both working chambers 65 and 67. When, however, the vehicle is cornering, the valve 36 produces a significant pressure differential between the chambers 65 and 67, which is maintained notwithstanding the damping action described above.

Figures 10 to 26 illustrate in more detail one arrangement for the main piston 54, the other components such as the valve pistons, damping pistons and springs have been omitted for clarity of illustration. In this arrangement, there are two of the valve bores 92 and two of the valve bores 94. In this arrangement, however, there is a single damping duct 116 and three damping ducts 118. This is chosen so as to provide more damping in a rebound action rather than in the opposite direction. As seen in Figures 10 and 11, the outer cylindrical surface of the main piston 54 is formed with slightly tapering end zones 150 and 152. Also it will be seen that there are four of the ducts 88 which open to a recess 154 in the groove 55 which thereby enables more uniform distribution of oil pressure within the groove 55 and therefore enable more uniform expansion of the seal 53. The recess 154 communicates with the ends of the ducts 88, as shown.

As shown in Figure 16, the damping duct 116 is formed in part by a stepped bore which includes an upper portion 160 and a lower wider portion 162. The step between the

two portions forms the valve seat 126 for the damping piston 120. The lower portion 162 is flanked by two side lateral bores 166 and 168. The lateral bores 166 and 168 provide the fluid communication paths to the lower face 68 of the main piston 54 once the damping piston 120 has been unseated. The parts of the lower portion 162 of the bore define the damping bore 122 which constrains the damping piston 120 to linear movement.

Each of the damping ducts 118 is formed in a similar way to the duct 116 and need not be described in detail. In each duct 118, the shoulder at the step in the duct 118 defines the valve seat 136 and the wider upper portion of the bore defines the second damping bore 132 which constrains the damping piston 130 to linear movement.

It is possible to arrange for the valve bores 92 and 94 to be provided with ports 138 at their upper ends and ports 144 at their lower ends so as to cooperate with laterally adjacent ducts 116 and 118.

Figure 26 shows the preferred shape for the upper washer 80. It will be seen that its outer edge 170 partly overlies each of the bores 92 and 94 and the duct 118 so as to allow fluid communication therewith. The outer edge 170 includes concave recesses 172 which overlie the bores 92 and 94, as shown so as to increase the exposed surface area of the pistons 96 and 130 to the hydraulic fluid in the working chamber 65.

Figure 27 illustrates in more detail the preferred shape for the lower washer 78. It has a generally round outer edge 174 which again partly overlies the bores 92 and 94 and the duct 116. Again, the edge 174 is provided with four concave recesses 176 to provide more exposure of the lower faces of the valve pistons 98 and 108 to the hydraulic fluid in the working chamber 67. Washers 80 and 78 may include dimples (not shown) which project somewhat inwardly relative to the ducts 118 and 116 so as to facilitate retaining of the ends of the damping springs 124 and 134 in the bores 122 and 132 respectively.

As indicated above, the damping performance of the ram is dependent on the spring force of the damping compression springs 124 and 134. In accordance with a preferred

embodiment of the invention, these can be selected to provide a predetermined damping response. For instance in the arrangement illustrated above, there are three of the damping ducts 118 and accordingly there are three of the damping compression springs 134. The compression forces of the springs 134 could be all the same but preferably they are arranged in increasing magnitude so that each of the damping pistons 130 will open at slightly different times as a pressure spike causes rapid increase of pressure within the cylinder. The springs 124 and 134 can also be of different strengths so that the suspension will have different responses to bumps and rebounds.

The dimensions of the device can be varied to suit requirements. Typically the diameter of the piston 54 is 32mm and the end zones 150 and 152 taper at say 5°. For automobiles and four wheel drive vehicles, it is expected that the diameter of the pistons 54 would normally be in the range from 35mm to 50mm. In this case the diameters of the pressure relief valve pistons and damping pistons would normally be in the range from 6mm to 9mm. The length of the piston 54 would be in the range of say 30mm to 35mm. The diameter of the bore 74 may be typically 10mm. The inner portions of the damping ducts may be of the order of 1mm less in diameter than the outer portions. The lengths of the pressure relief pistons and damping pistons would normally be in the range 5mm to 10mm. It is thought that the weight of the pistons is not critical because the hydraulic fluid pressures to which they will be subjected will be very substantial and will tend to negate effects caused by inertia of the pistons. The total length of the main piston 54 may be say 30mm to 35mm. Typically the valve pistons 96, 98, 106 and 108 will need to move through a relatively small distance, say 2mm, before the ports associated therewith are opened. The total stroke of the main piston 54 is of the order of 150mm. The springs 100 and 110 can be very light weight since their sole function is to keep the pressure relief pistons 96, 98 and 106, 108 from sticking together and blocking the duct 86. The damping springs 124 and 134, however, do have an important effect on the performance of the ram because they change the damping rate. The values of these springs, however, can be varied considerably from very light weight which will produce a very soft shock absorber to comparatively heavy weight which will produce a stiff shock absorber. Typically the spring constant of these springs will be in the range from about 0.1kg/cm to 55kg/cm but

these may be varied outside this range in special circumstances. For heavier vehicles such as trucks and the like, the diameter of the piston 54 would be greater, say in the range from 50mm to 100mm. The diameters of the pressure relief valve pistons and damping pistons would normally be in the range from 7mm to 12mm. For heavier vehicles, the springs 100 and 110 can be light but the damping springs 124 and 134 will be correspondingly heavier.

Figure 28 diagrammatically illustrates an adjustable shock absorber 250 constructed in accordance with the invention. The shock absorber essentially comprises an hydraulic ram constructed in accordance with the rams described and illustrated above and therefore details of the piston thereof need not be described. Its couplings 58 and 60 are connected to lines 252 and 254 which are in fluid communication with an hydraulic fluid reservoir 256. The coupling 84 is connected to a control line 258 which receives control signals in the form of pressure variations in the control fluid therein. These control signals could be derived from any device which produces variable pressure signals in response to a manual or other input. For instance the device could be like the control valve 36 except that the arm 40 is manually operated in order to increase or decrease the stiffness of the shock absorber 150 as required. Increasing the pressure in the control line 258 would have the effect of requiring larger pressure spikes in the fluid above and below the piston 54 in order to move the auxiliary pistons to the point wherein they open to allow movement of the main piston 54. In other words, the stiffness of the shock absorber 250 can be increased or decreased by increasing or decreasing pressure on the control line 258.

Many modifications will be apparent to those skilled in the art without departing from the spirit and scope of the invention.

CLAIMS

1. An active suspension system (12) for a vehicle (2) including at least one fluid operated ram (14,16,18,20) having a cylinder (52) and a main piston (54) mounted therein for reciprocating movement, control means (36,38,40) for controlling an equilibrium position of the piston in the ram in response to lateral acceleration of the vehicle in order to counter the effects of body roll of the vehicle, and wherein the ram includes shock absorbing means (70) for permitting rapid movement of the piston from said equilibrium position when operating fluid in the ram is subjected to transient increases in pressure.

2. An active suspension system as claimed in claim 1 wherein said at least one ram includes coupling means (62,64) to couple the ram between the vehicle chassis (22) and an axle assembly (24) thereof and wherein said at least one ram constitutes a shock absorber for the suspension system.

3. An active suspension system as claimed in claim 2 wherein the vehicle has a plurality of wheels each being connected to the vehicle chassis (22) by one of said axle assemblies (24) and wherein one of said rams is coupled between the vehicle chassis and respective axle assemblies.

4. An active suspension system as claimed in claim 3 wherein the vehicle is an automobile having front and rear driver side wheels and front and rear passenger side wheels and wherein there are front and rear driver side rams and front and rear passenger side rams and wherein the control means is operable to cause the driver side rams to simultaneously move to a first equilibrium position in one sense when the vehicle is cornering in one direction and to cause the passenger side rams to simultaneously move to a second equilibrium position in another sense, opposite to said one sense, when the vehicle is cornering in another direction, opposite to said one direction.

5. An active suspension system as claimed in claim 4 wherein the shock absorbing means of each of said rams is independently operable.

6. An active suspension system as claimed in any one of claims 1 to 5 wherein for each ram, first and second working chambers (65,67) are defined in the cylinder on opposite sides of said piston and wherein the piston includes at least one damping passage (116,118) therethrough and valve means (96,98,106,108,120,130) to control flow of operating fluid through said damping passage when said fluid in one or other of said working chambers is subjected to transient increases in pressure.

7. An active suspension system as claimed in claim 6 wherein said valve means includes at least a valve bore (92,94) which extends from first and second end faces (66,68) of said main piston and first and second valve pistons (96,98) mounted for reciprocating movement in said at least one bore and wherein said main piston is mounted on a piston rod (56) which includes a fluid control duct (92) which receives operating fluid from said control means and wherein the main piston includes ducts (86) which provide a fluid path from said fluid control duct to said valve bore.

8. An active suspension system as claimed in claim 7 wherein, in the absence of transient increases in pressure in said working chambers, the damping passage is closed and the control means is operable to generate pressure differentials in said working chambers to cause movement of the main piston to equilibrium positions which are dependent on lateral acceleration of the vehicle and to maintain said pressure differentials in order to maintain the equilibrium positions which have been obtained, and wherein the control means is arranged to supply to said control fluid duct the higher of the pressures supplied to the working chambers.

9. An active suspension system as claimed in claim 8 wherein, if a transient increase in pressure occurs in the fluid in either of said working chambers the first or second valve piston is displaced to thereby open a port (138,144) in said at least one damping passage, whilst the pressure differentials in said working chambers are maintained.

10. An active suspension system as claimed in claim 9 wherein said at least one damping passage includes a damping valve (120,126,130,136) which opens once subjected to a predetermined pressure differential thereacross.

11. An active suspension system as claimed in claim 10 wherein the damping valve includes a damping piston (120,130) which is biased into engagement with a valve seat (126,136) by means of a damping spring (124,134) and wherein the damping piston is unseated from the valve seat when a transient increase of pressure in said working fluid occurs.

12. An active suspension system as claimed in claim 11 wherein there are at least two of said damping passages each having one of said damping valves therein to permit flow of fluid therethrough in opposite directions depending on whether the transient increase in pressure is in the first or second working chamber.

13. An hydraulic ram (14,16,18,20) which can be used as an adjustable shock absorber, said ram having a cylinder (52), a main piston (54) mounted on a piston rod (56) and defining first and second working chambers (65,67) in the cylinder;

at least one first damping passage (116) extending through the main piston and having a first damping valve (120,126) therein for permitting flow of operating fluid from the first working chamber to the second working chamber;

at least one second damping passage (118) extending through the main piston and having a second damping valve (130,136) therein for permitting flow of operating fluid from the second working chamber to the first working chamber;

first and second transient pressure relief valves (96,138;108,144) in said first and second damping passages respectively;

the first transient relief valve (96,138) including a first transient relief piston (96) mounted for sliding movement in a first bore (92) in said main piston, said first relief piston (96) having a first end which is in fluid communication with said first working chamber;

the second transient relief valve (108.144) including a second transient relief piston (108) mounted for sliding movement in a second bore (94) in said main piston. said second relief piston (108) having a first end which is in fluid communication with the second working chamber; and

a control fluid duct (82) in said piston rod (56) in communication with passages (86) in the main piston to provide fluid communication between said fluid control duct and said first and second bores whereby second ends of the first and second transient relief pistons are subjected to the fluid pressure applied to said fluid control duct. and wherein the arrangement is such that hydraulic fluid supply means can be used to supply operating fluid at a high pressure and a low pressure to said first and second working chambers whereby the pressure difference across the main piston enables the ram to support a load and wherein the high pressure is, in use. applied to said control fluid duct so that the first and second transient relief pistons are closed but, if a transient increase of fluid pressure is caused in the first or second working chamber, the first or second transient relief pistons will be transiently displaced to permit flow of fluid through the first or second damping passage respectively.

14. A ram as claimed in claim 13 wherein the first and second damping passages include first and second damping valves (120,124,126; 130,134,136) respectively.

15. A ram as claimed in claim 14 wherein the first damping valve (120,124,126) includes a first damping piston (120) mounted for reciprocating movement in the main piston and being biased into engagement with a first valve seat (126) by means of a first damping spring (124) the arrangement being such that the first damping piston will be unseated from said first valve seat when the pressure in the first damping passage reaches a first predetermined level following opening of the first transient pressure relief valve; and wherein the second damping valve (130,134,136) includes a second damping piston (130) mounted for reciprocating movement in the main position and being biased into engagement with a second valve seat (136) by means of a second damping spring (134). the arrangement being such that the second damping piston will be unseated from said

second valve seat when the pressure in the second damping passage reaches a second predetermined level following opening of the second transient relief valve.

16. A ram as claimed in claim 15 wherein the main piston has a plurality of said first and second damping passages each having one or more of said pressure relief valves associated therewith.

17. A vehicle (2) having an active suspension (12) as defined in any one of claims 1 to 12.

18. A vehicle (2) having an active suspension (12) as defined in claim 1 and wherein the fluid operated ram thereof comprises a ram as defined in any one of claims 13 to 16.

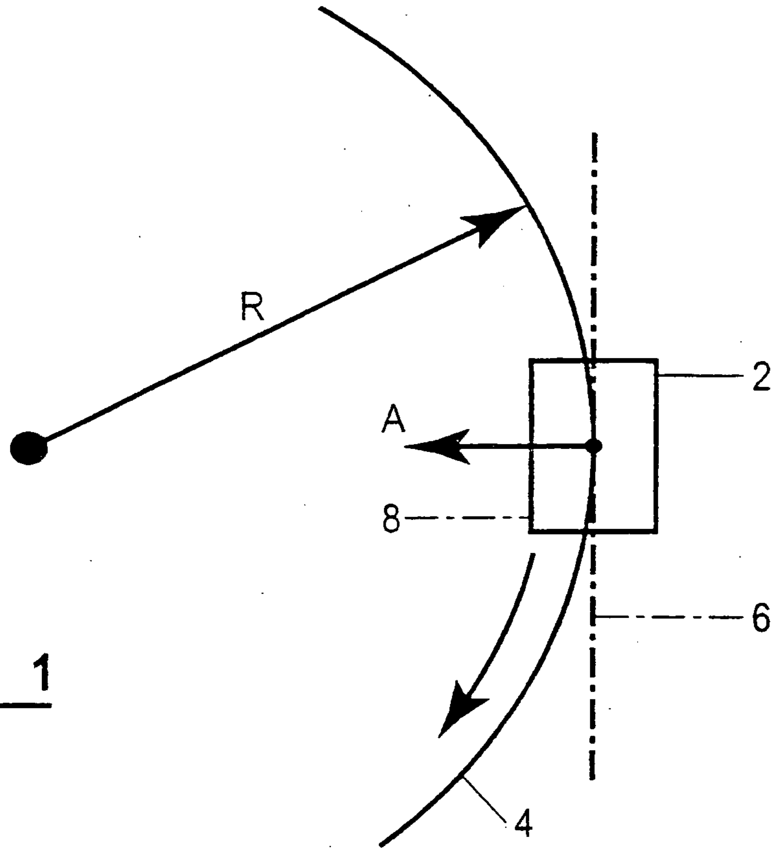

FIG 1

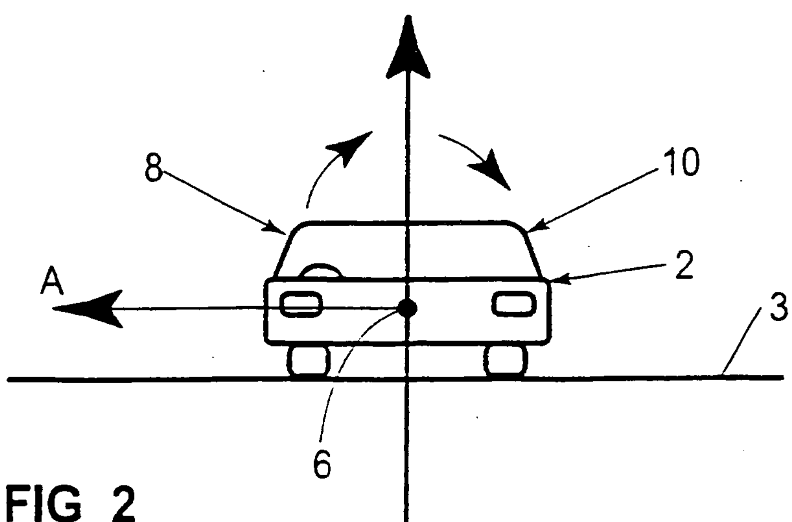

FIG 2

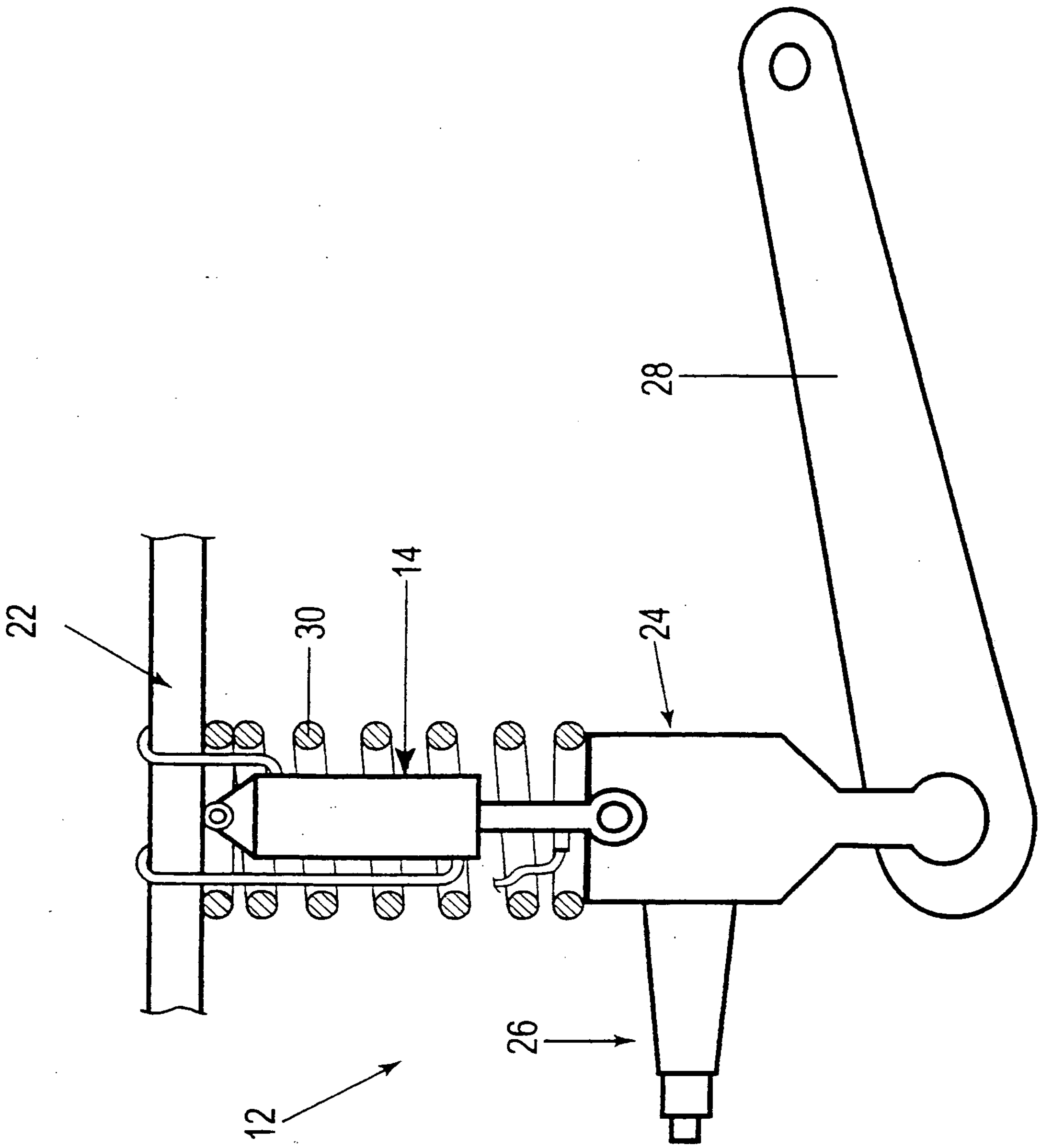

FIG 3

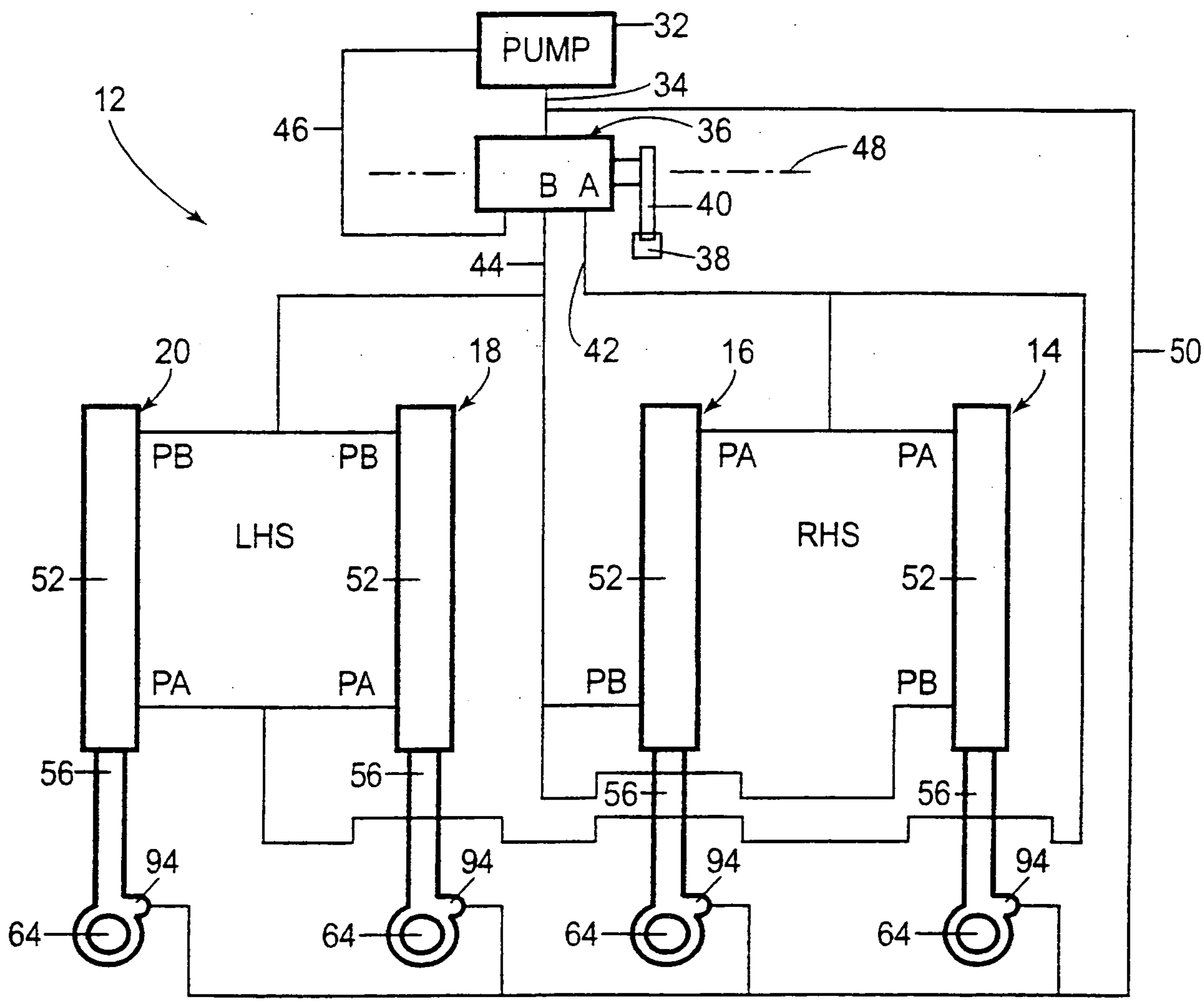
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FIG 4

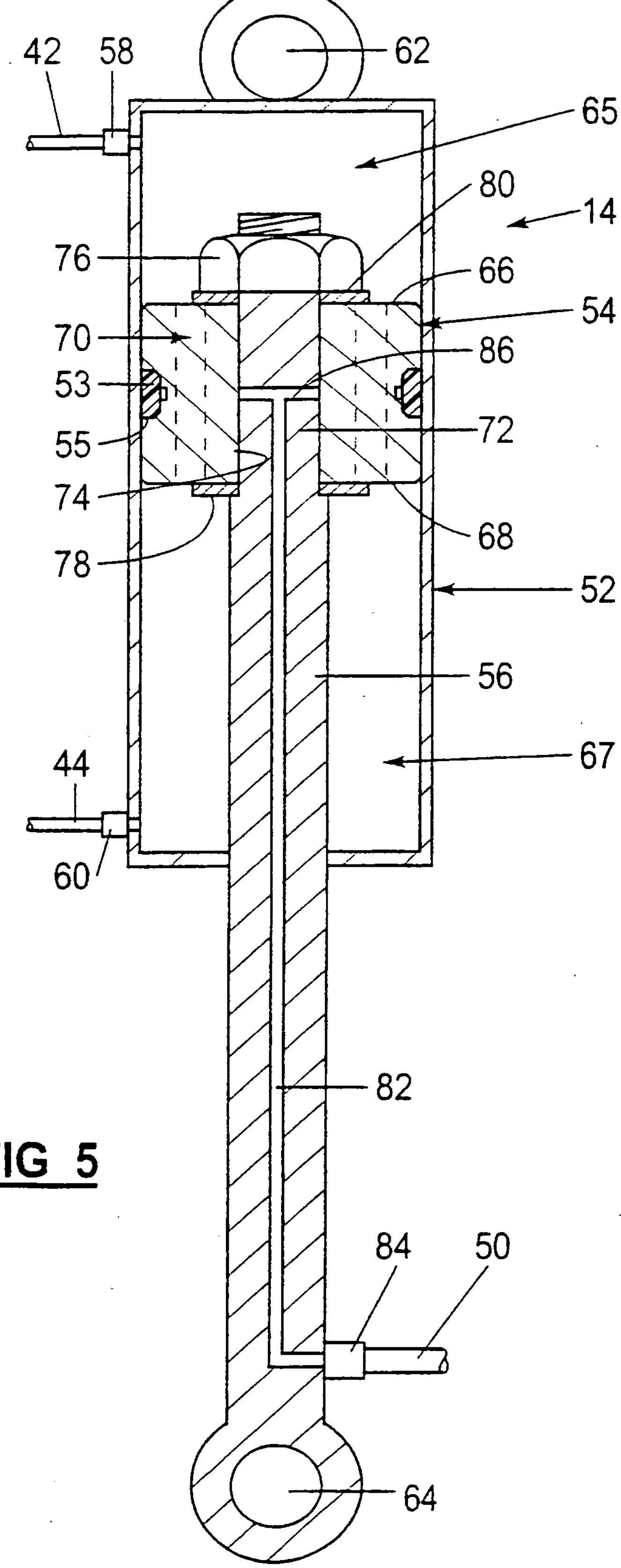
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FIG 5

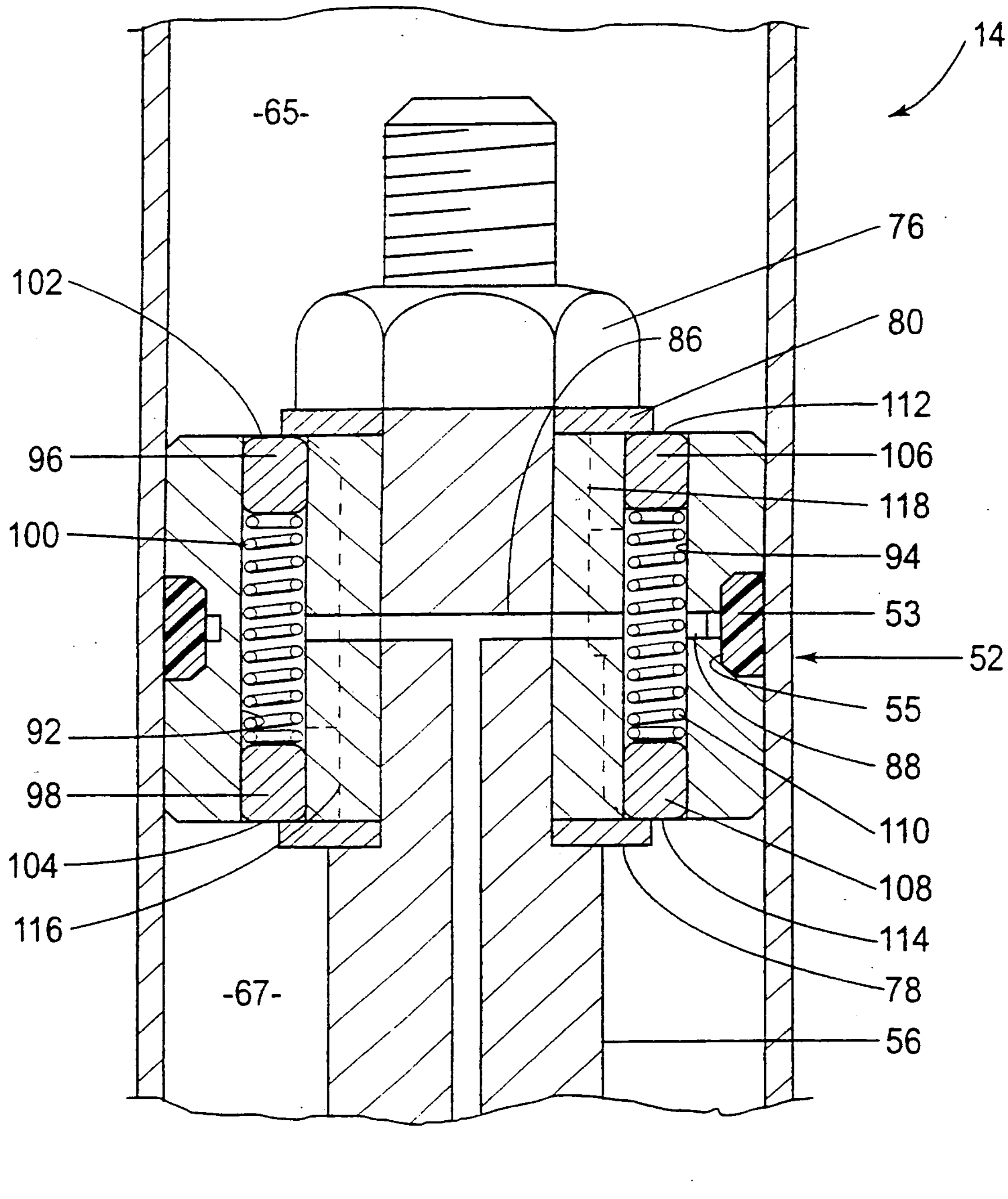
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FIG 6

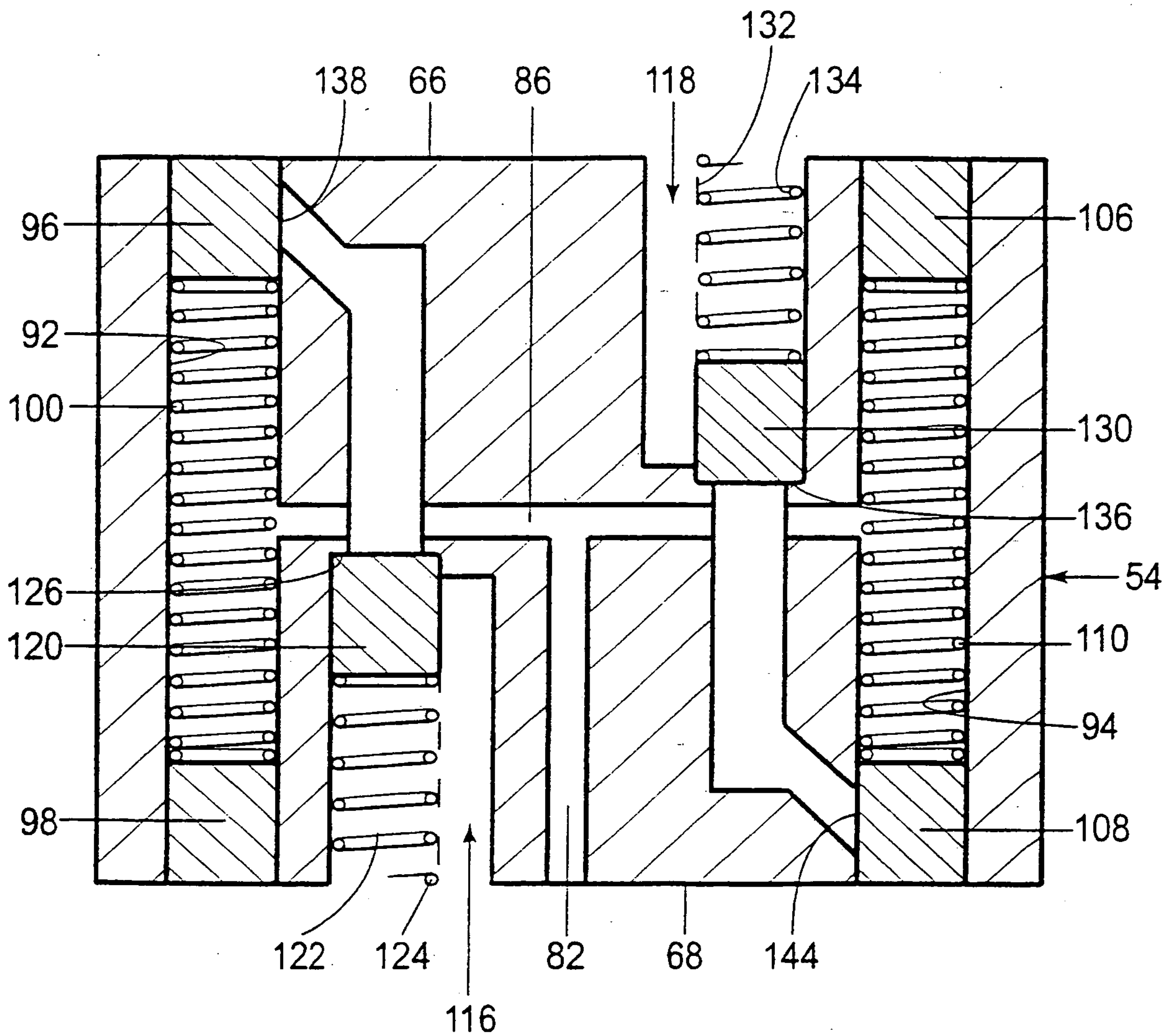

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FIG 7

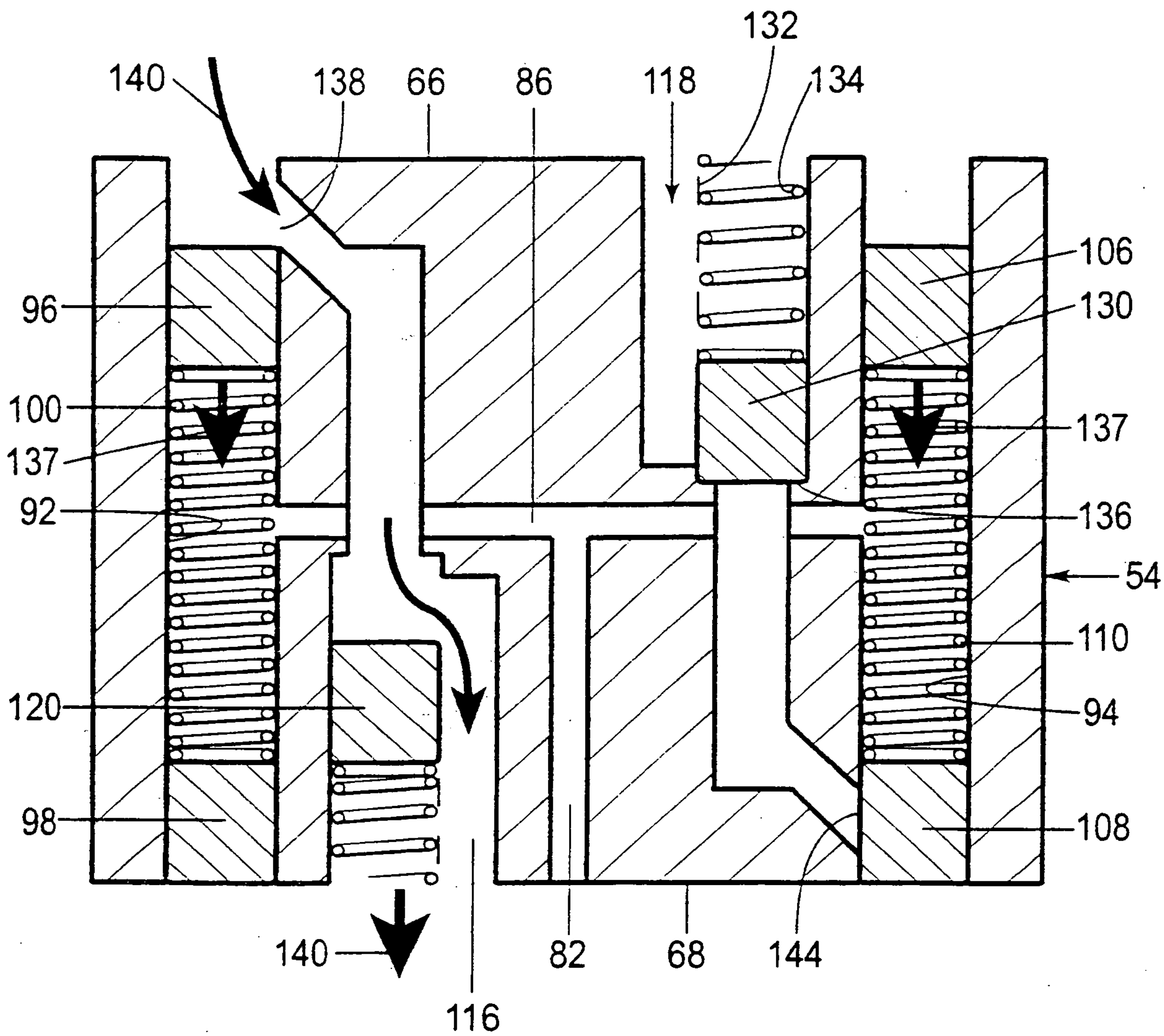
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FIG 8

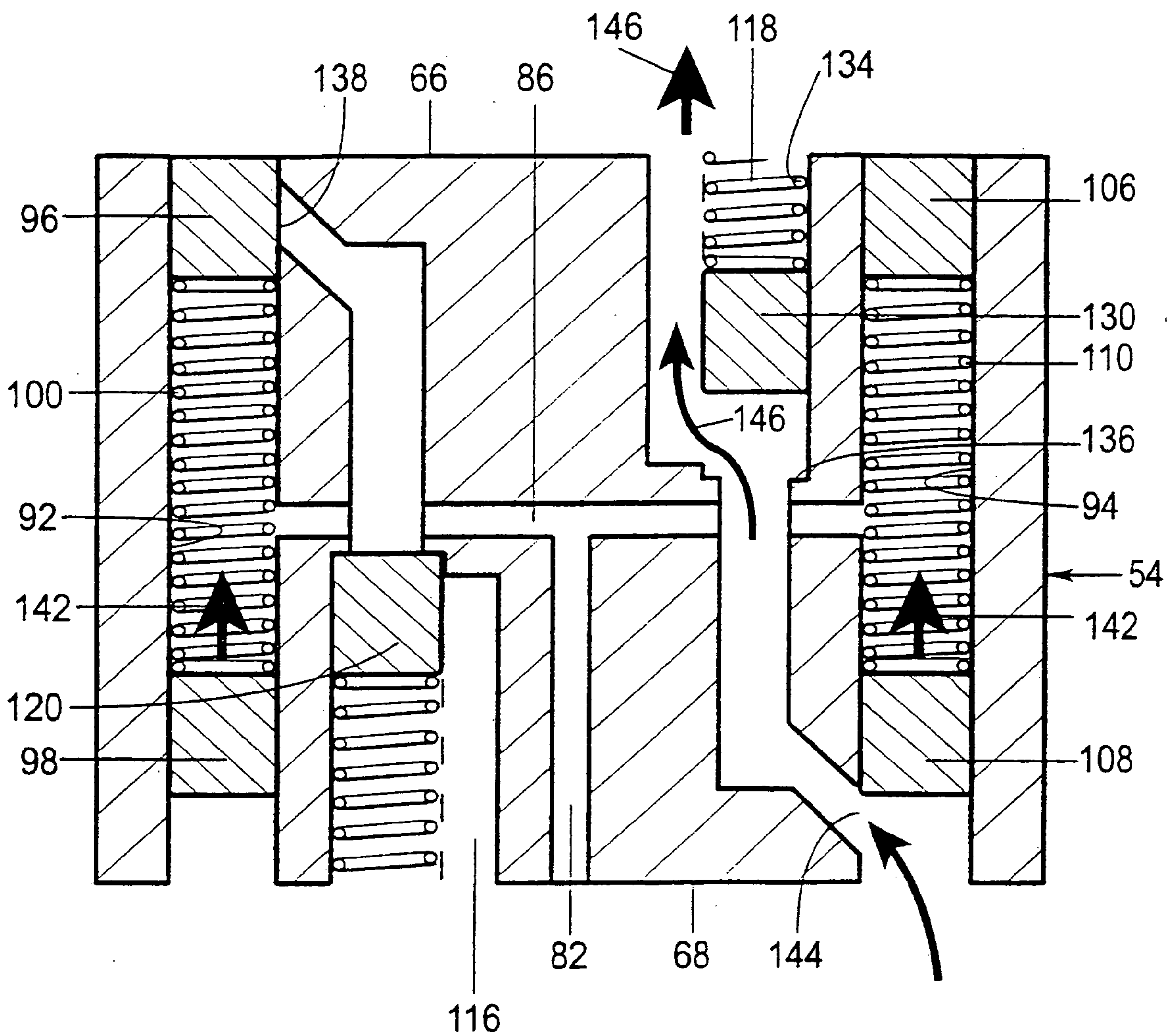
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FIG 9

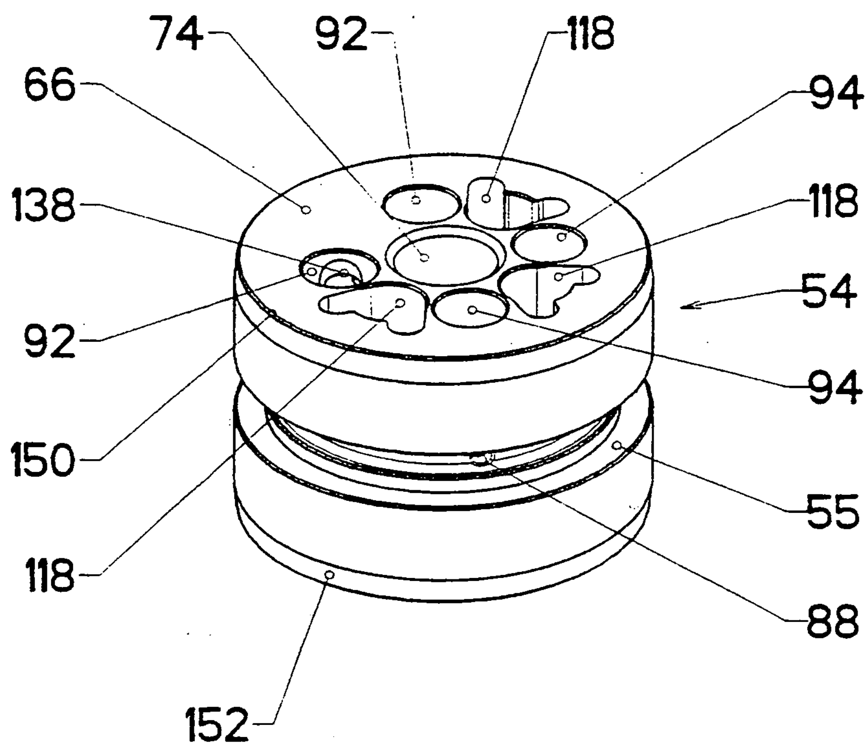

FIG 10

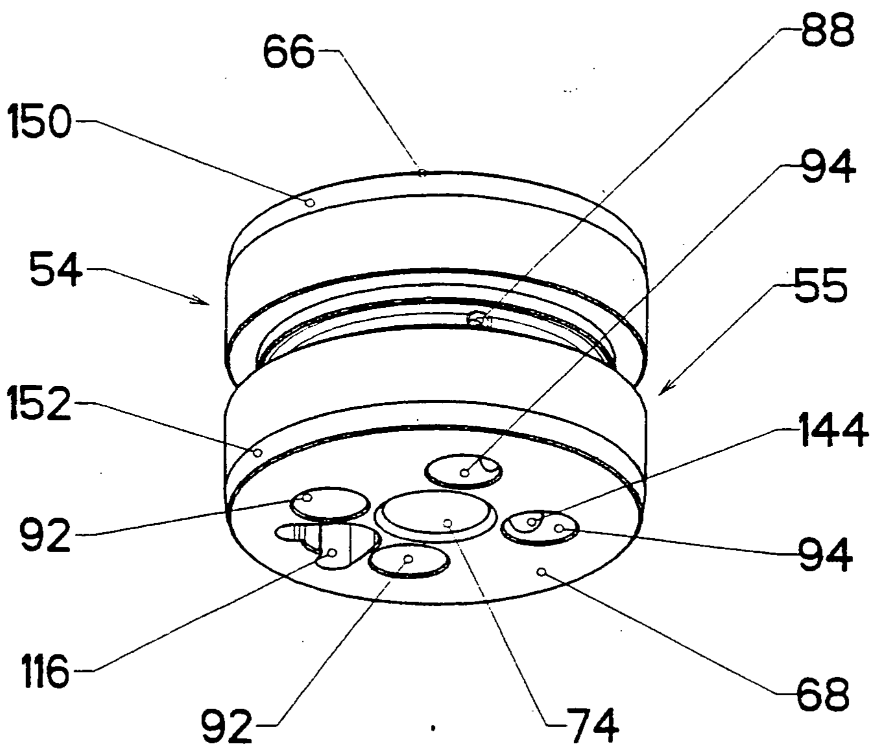

FIG 11

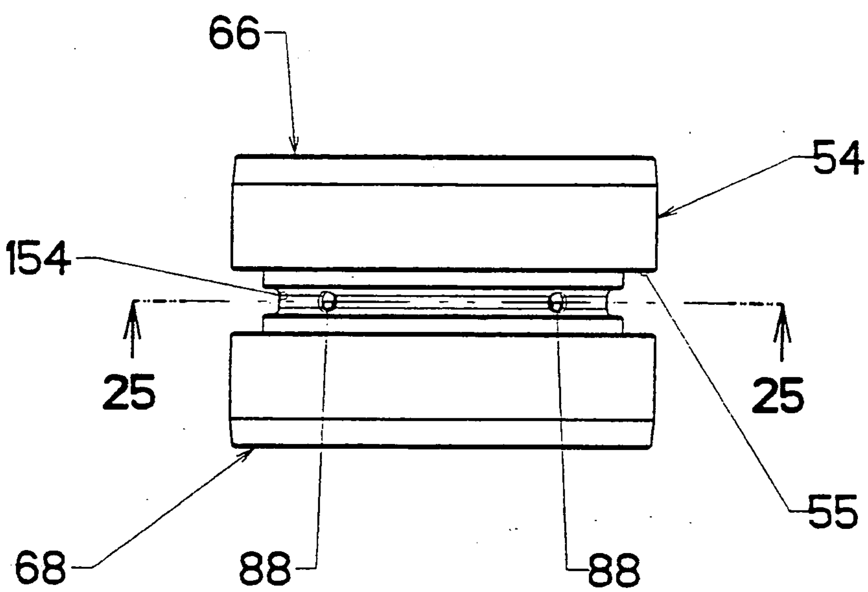

FIG 12

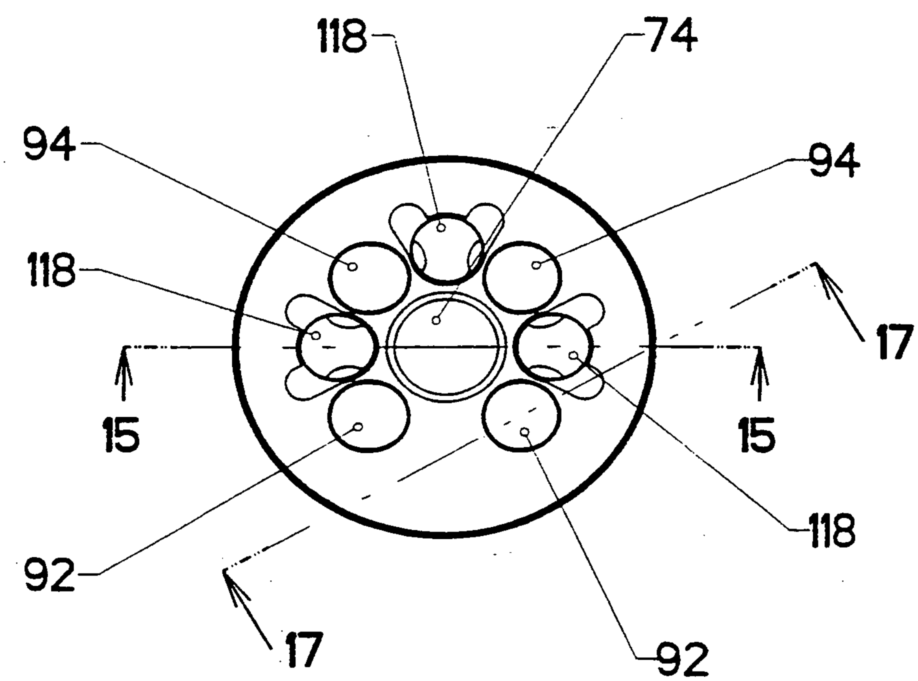

FIG 13

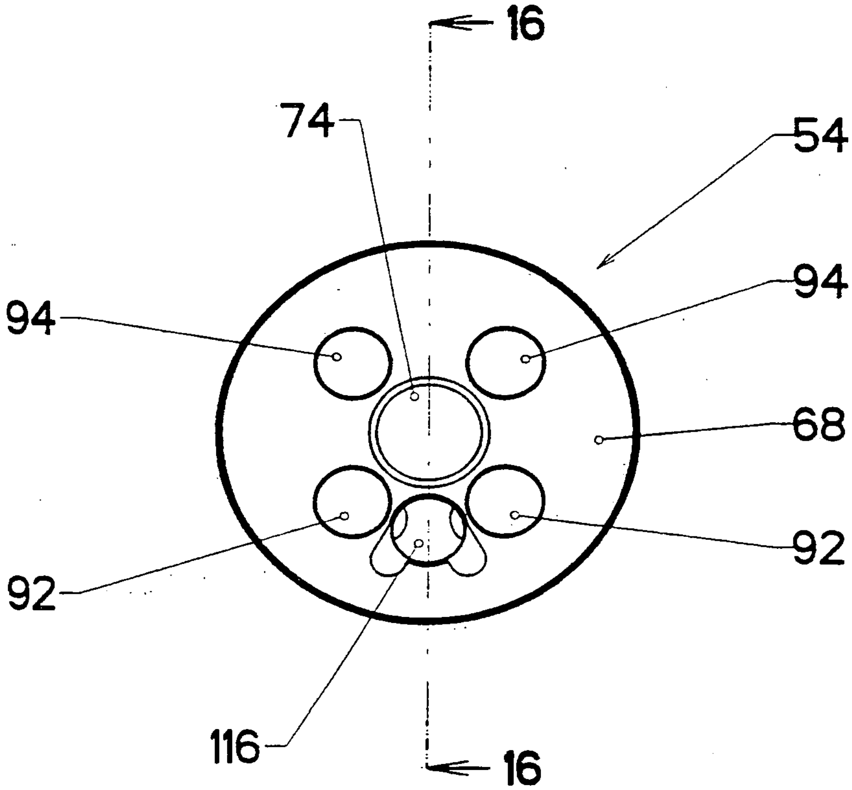

FIG 14

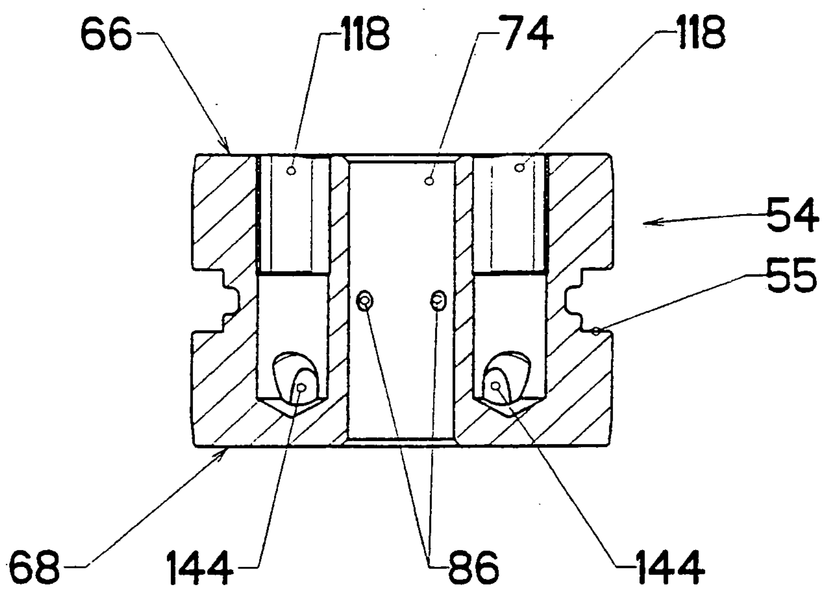

FIG 15
SECTION 15-15

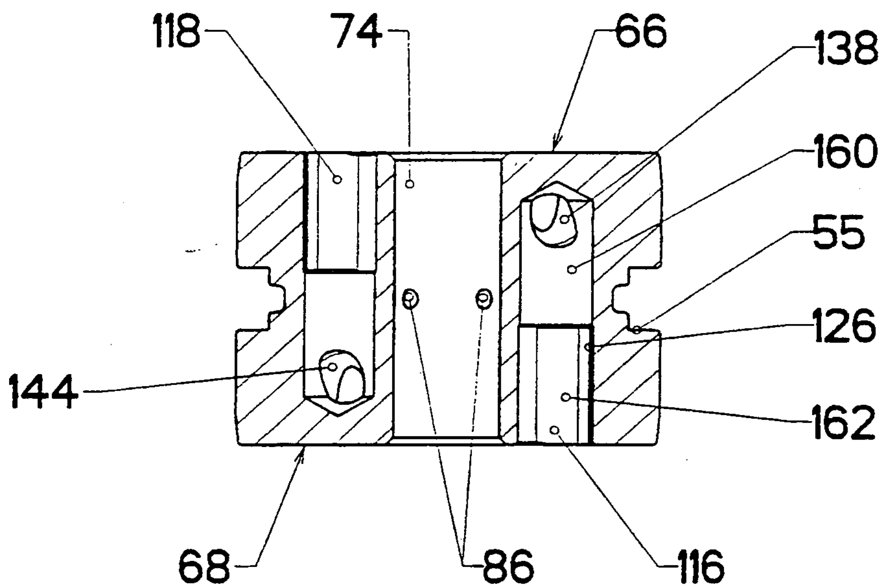

FIG 16
SECTION 16-16

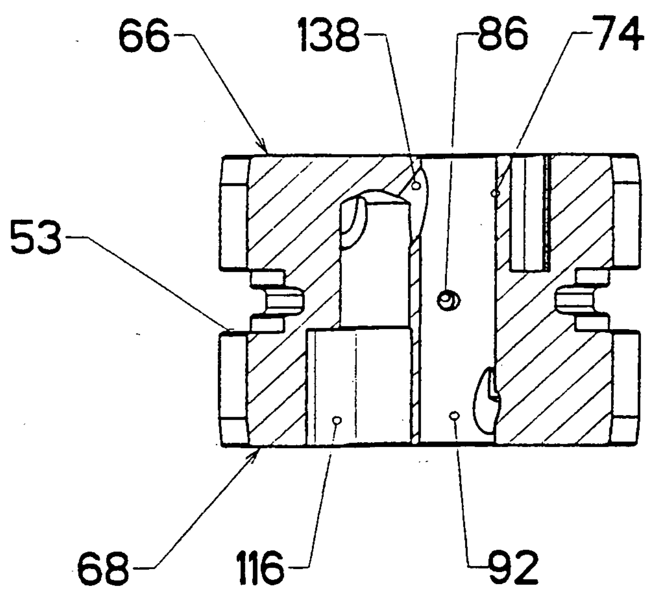

FIG 17
SECTION 17-17

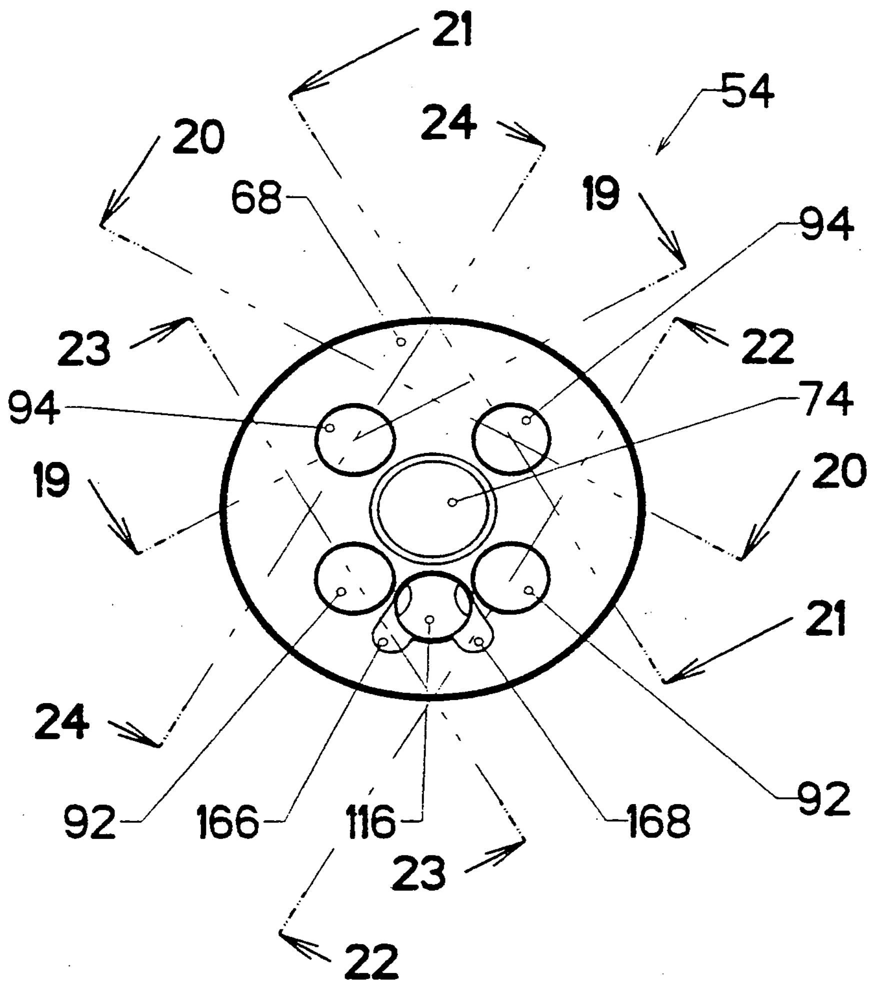

FIG 18

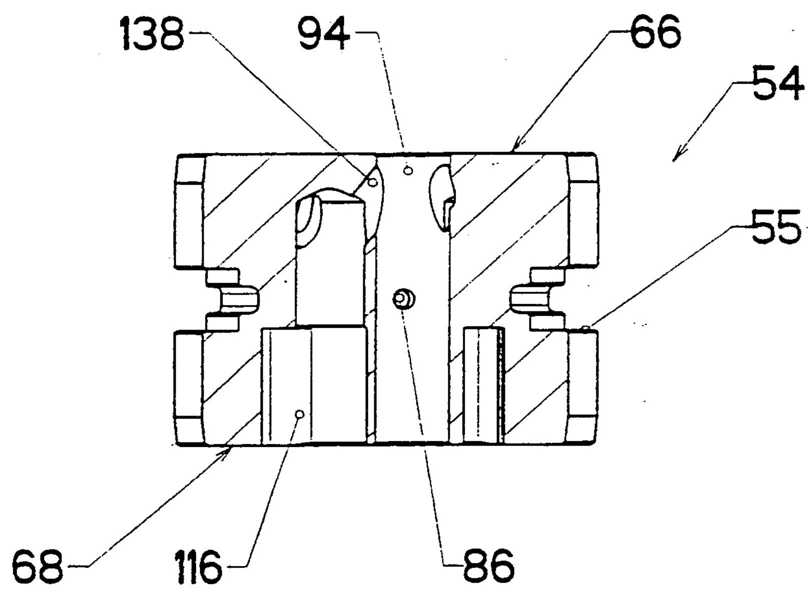

FIG 19
SECTION 19-19

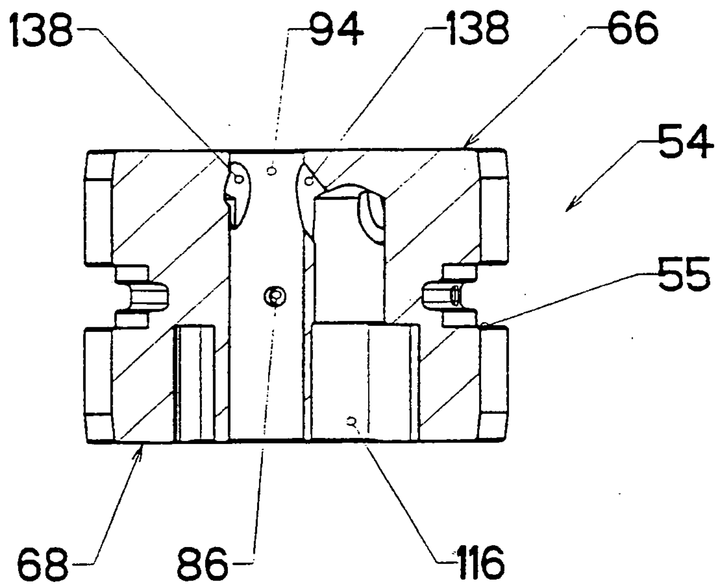

FIG 20
SECTION 20-20

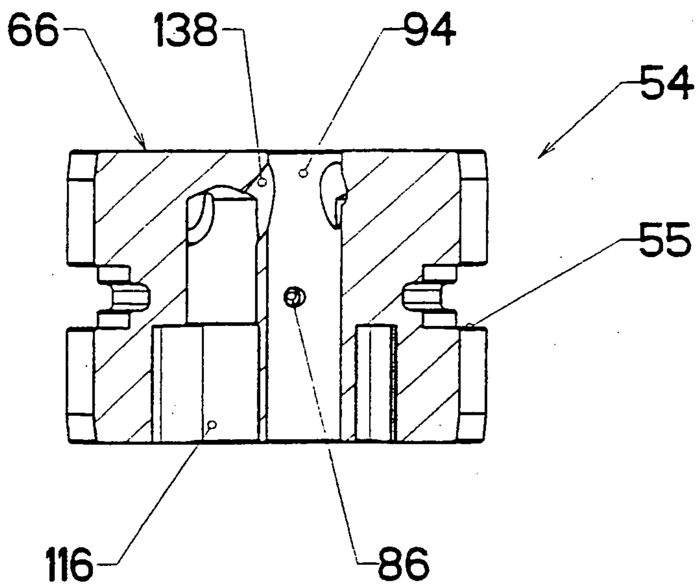

FIG 21
SECTION 21-21

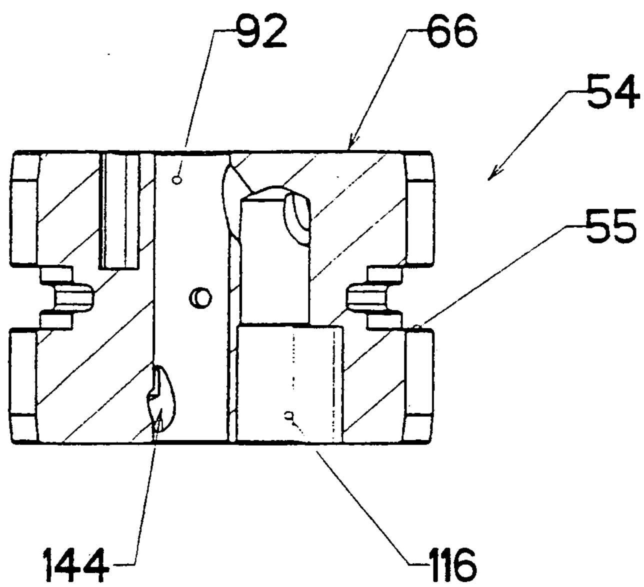

FIG 22
SECTION 22-22

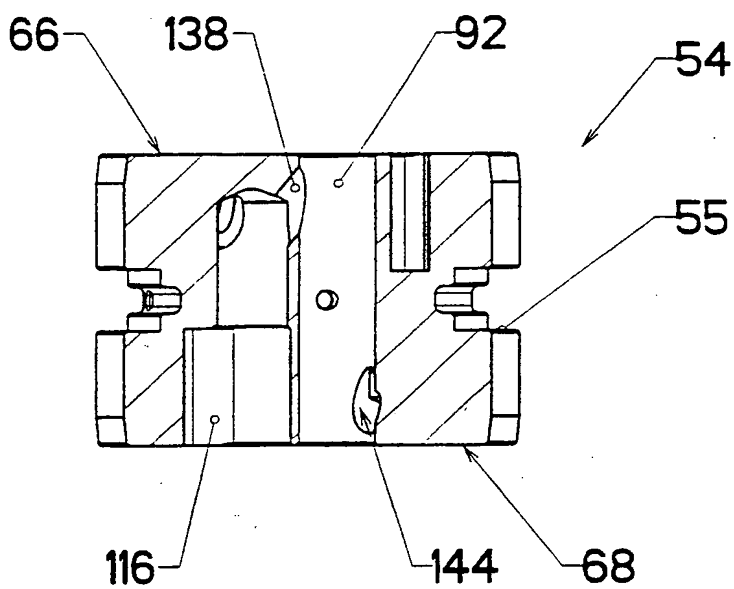

FIG 23
SECTION 23-23

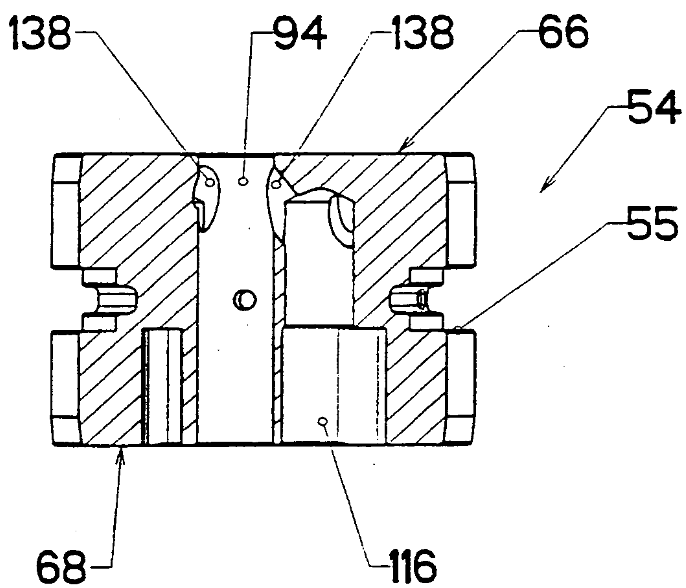

FIG 24
SECTION 24-24

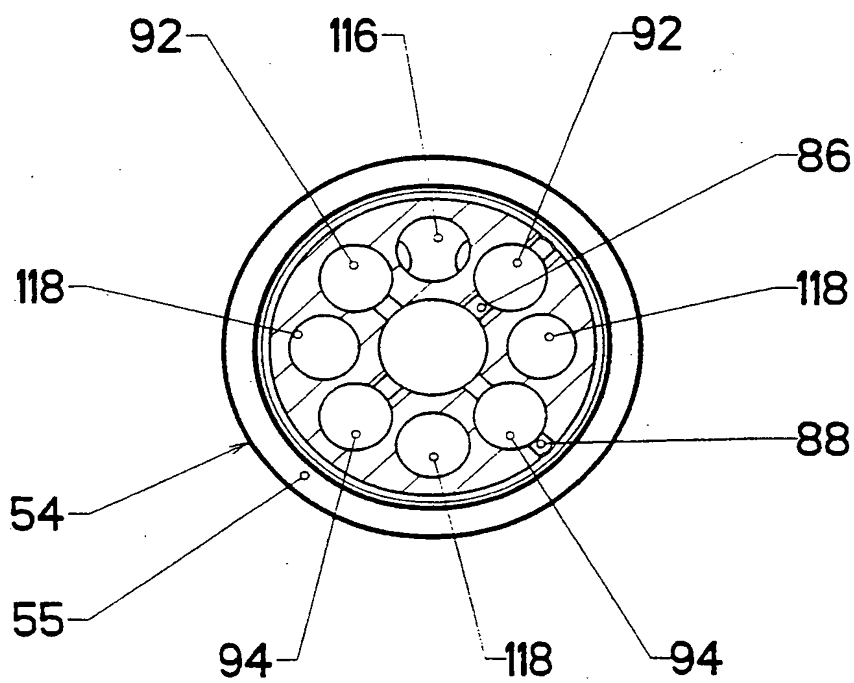

FIG 25
SECTION 25-25

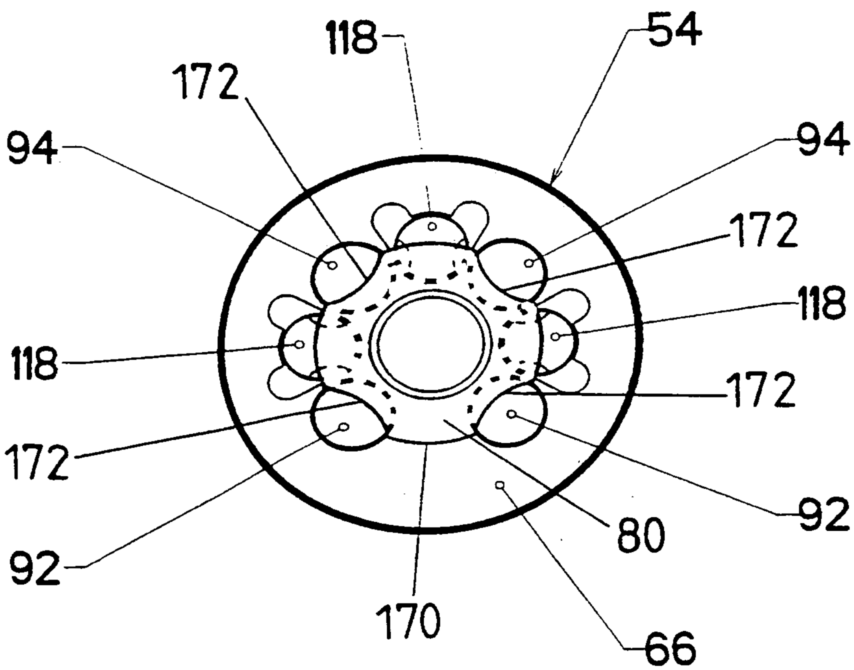

FIG 26

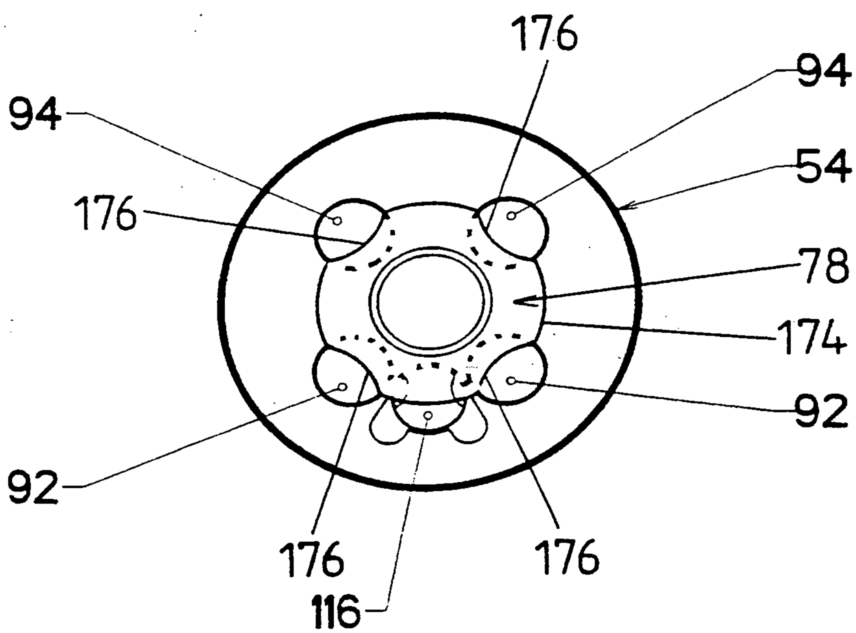

FIG 27

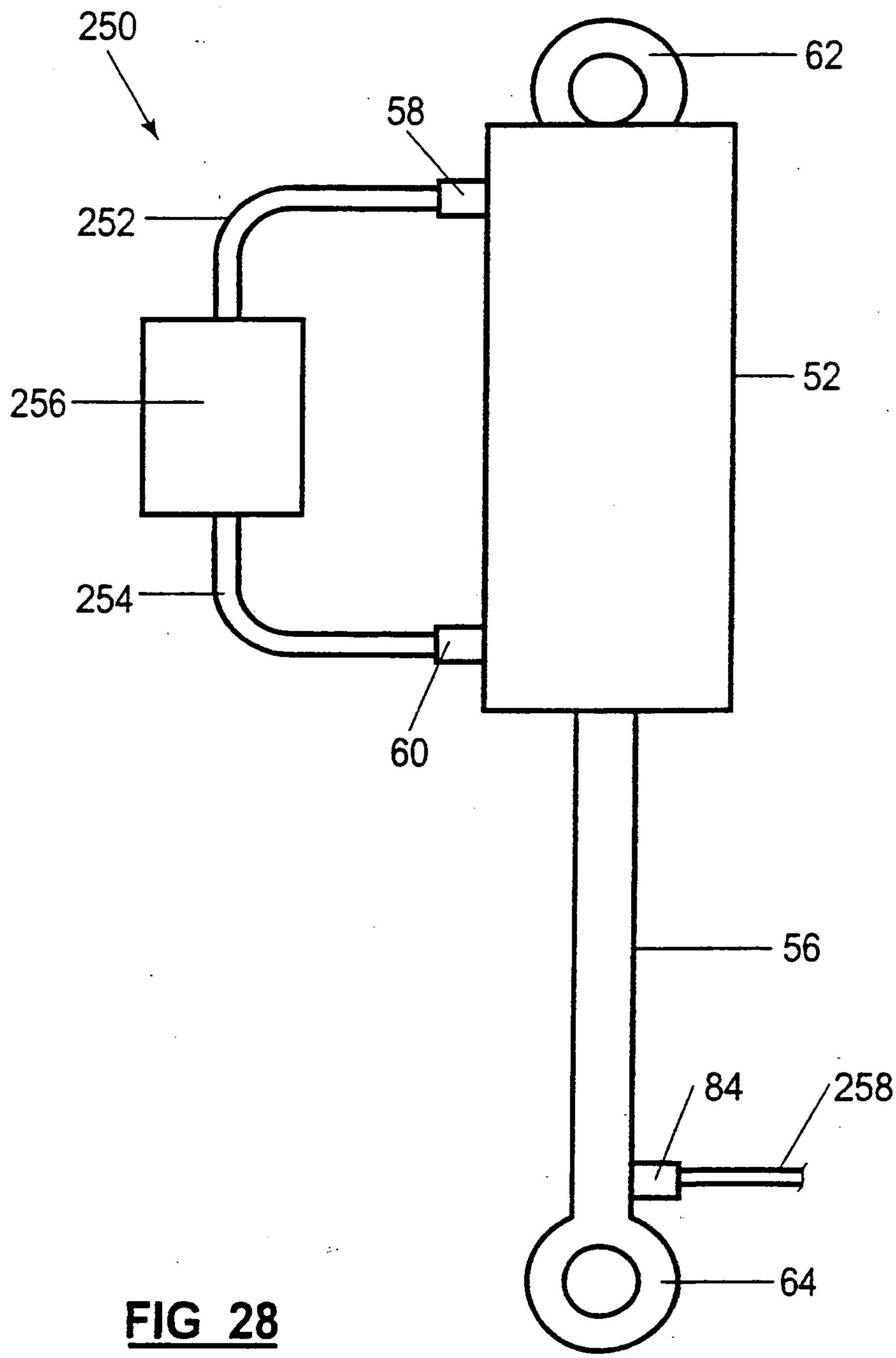

FIG 28

INTERNATIONAL SEARCH REPORT

International application No.
PCT/AU00/00470

A. CLASSIFICATION OF SUBJECT MATTER

Int. Cl. 7: B60G 17/01, 17/08, 15/12; F16F 9/19, 9/504

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
IPC: B60G 17/01, 17/08; 15/12; F16F 9/19, 9/504

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched
AU: IPC AS ABOVE

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)
WPAT: Keyword search

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	US 5653315 A (EKQUIST et al) 5 August 1997 Column 1 line 30 to column 6 line 41, column 10 line 33 to column 12 line 58, claims 7, 8 & 18, and Figures 2- 7	1-18
X	US 5123671 A (DRIESSEN et al) 23 June 1992 Column 1 line 20 to column 12 line 61, claims 18, 19 - 21, 30 - 36, 49, 58 & 74 and Figures 4a, 6a, & 15 - 19	1-18
X	US 4890858 A (BLANKENSHIP) 2 January 1990 Column 1 line 29 to column 150 line 18, claims 1, 14, 25 & 36 And Figures 3 - 7	1-18

[X] Further documents are listed in the continuation of Box C [X] See patent family annex

* Special categories of cited documents:

"A" document defining the general state of the art which is not considered to be of particular relevance
"E" earlier application or patent but published on or after the international filing date
"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)
"O" document referring to an oral disclosure, use, exhibition or other means
"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention
"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone
"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art
"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
21 June 2000	29 JUN 2000
Name and mailing address of the ISA/AU AUSTRALIAN PATENT OFFICE PO BOX 200, WODEN ACT 2606, AUSTRALIA E-mail address: pct@ipaustralia.gov.au Facsimile No. (02) 6285 3929	Authorized officer **LIONEL BOPAGE** Telephone No : (02) 6283 2153

C (Continuation).	DOCUMENTS CONSIDERED TO BE RELEVANT	
Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	US 5092626 A (ATHANAS et al) 3 March 1992 Column 1 line 19 to column 14 line 24, claims 1 - 19 and figure 15	1-18
X	US 4867475 A (GROVES) 19 September 1989 Column 1 line 12 to column 24 line 34, claims 10 - 19 and figures 2 - 7	1-18

INTERNATIONAL SEARCH REPORT Information on patent family members	International application No. **PCT/AU00/00470**

This Annex lists the known "A" publication level patent family members relating to the patent documents cited in the above-mentioned international search report. The Australian Patent Office is in no way liable for these particulars which are merely given for the purpose of information.

Patent Document Cited in Search Report		Patent Family Member					
US	5653315	DE	19510092	GB	2287769	JP	7257133
US	5123671	DE	4005601	GB	2229253	JP	2240425
		US	5016908	US	5092626		
US	5092626	DE	4005601	GB	2229253	JP	2240425
		US	5016908	US	5123671		

END OF ANNEX